Tuning of Coordinated Controllers for Boiler-turbine Units1)

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Abstract A simple two-by-two model for a boiler-turbine unit is derived in this paper, which can capture the essential dynamics of the typical process. The coordinated controller is designed based on this model. A PID control structure and a tuning procedure are proposed. The example shows that the method is easy to use and can achieve an acceptable performance.

Key words Boiler-turbine unit, PID tuning, robustness, industrial applications

1 Introduction

The coordinated controller design for a boiler-turbine unit has attracted much attention in the past years. Modern control techniques, *e.g.*, generalized predicative control (GPC)[1], LQG/LTR[2] and H_∞ control[3~4], have been used to improve the unit performance. These results are encouraging, however, the final controllers are usually of very high orders. For example, the order of the H_∞ controller in [4] is 15 even after reduction, which is hardly possible to implement in practice. Further, the structures of the controllers are complex and the parameters are hard to tune when needed.

Recently, the second author[5] of this paper proposed a PID reduction procedure for a multivariable controller and showed that the performance of the final multivariable PI controller for a boiler-turbine unit did not significantly degenerate from the original loop-shaping H_∞ controller. Encouraged by this result we will study PID tuning for a boiler-turbine unit in this paper.

The rest part of the paper is arranged as follows: In Section 2 we will first derive a simple model for a boiler-turbine unit. Then in Section 3 we will find a control structure for this model and propose a method to tune the controller parameters. An example is given in Section 4 to illustrate the proposed tuning method, and conclusions and further research topics are given in Section 5. Throughout this paper, we will use $\triangle$ to denote the increment of a variable.

2 Simple boiler-turbine model

To study controller tuning for a boiler-turbine unit, it is helpful to find a simple model that can capture the essential dynamics, especially the coupling effect between the generated electricity and the throttle pressure. Cheres[6] and de Mello[7] proposed a nonlinear dynamic system with a simple structure to capture the essential dynamics of the boiler-turbine unit (Fig. 1).

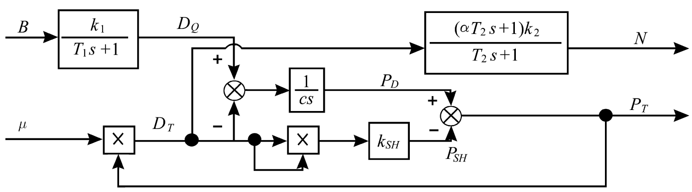

Fig. 1 A simple diagram of a boiler-turbine unit

The model in Fig. 1 shows the energy balance relation and the essential nonlinear characteristics of the boiler-turbine unit.

The energy balance relation: the drum pressure P_D shows the balance between the heat flux from

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furnace D_Q and the turbine steam flow D_T, *i.e.*,

$$\triangle D_Q - \triangle D_T = c\frac{\mathrm{d}\triangle P_D}{\mathrm{d}t} \tag{1}$$

The nonlinear characteristics are

1) the pressure drop P_{SH} between the drum pressure P_D and the steam pressure P_T is related with outlet steam flow D_T, which can be described by

$$P_{SH} = P_D - P_T = k_{SH}D_T^2 \tag{2}$$

2) the steam flow D_T is the product of the main pressure P_T and the turbine governor position μ, *i.e.*,

$$D_T = \mu P_T$$

Table 1 Nomenclature

B	Boiler firing rate
μ	Governor valve position
N	Electricity generated
P_T	Throttle pressure
P_D	Drum pressure
P_{SH}	Pressure drop between drum and main steam valve
D_Q	Heat flux from furnace
D_T	Turbine steam flow
c	Boiler and tubes storage constant
k_{SH}	Superheater friction drop coefficient
α	Power coefficient of HP cylinder
T_1	Combustion and heat transfer constant
T_2	Reheater time constant

Consider the behavior of the boiler-turbine unit at the nominal condition. Take the increments on both sides of (2) and (3), we have

$$\triangle P_D - \triangle P_T = R\triangle D_T \tag{4}$$

$$\triangle D_T = \mu\triangle P_T + P_T\triangle\mu \tag{5}$$

where $R = 2k_{SH}D_T$ is the friction of tubes. Combining (1), (4) and (5) we have

$$\triangle D_T = \frac{1}{T_0 s+1}\triangle D_Q + \frac{P_T T_t s}{T_0 s+1}\triangle\mu \tag{6}$$

$$\triangle P_T = \frac{1}{\mu(T_0 s+1)}\triangle D_Q - \frac{P_T}{\mu}\frac{T_b s+1}{T_0 s+1}\triangle\mu \tag{7}$$

where

$$T_0 := (R+\frac{1}{\mu})c, \quad T_b := Rc, \quad T_t := \frac{c}{\mu} \tag{8}$$

From Fig. 1, we see the transfer function from the boiler firing rate B to the steam flow rate D_Q can be approximated by a first-order model, *i.e.*,

$$\triangle D_Q = \frac{k_1}{T_1 s+1}\triangle B \tag{9}$$

and the transfer function of power generation of a turbine can be written as

$$\triangle N = \frac{(aT_2 s+1)k_2}{T_2 s+1}\triangle D_T \tag{10}$$

where α is the ratio of the electric power generated by the high-pressure turbine to the total electric power generated by the turbine.

Combining (6), (7), (9) and (10), we obtain the linear model for the boiler-turbine unit at a certain operating point as follows.

$$\begin{bmatrix} \triangle P_T \\ \triangle N \end{bmatrix} = T(s)G_0(s)B(s)\begin{bmatrix} \triangle B \\ \triangle \mu \end{bmatrix} \tag{11}$$

where

$$T(s) = \begin{bmatrix} 1 & 0 \\ 0 & \dfrac{(aT_2s+1)k_2}{T_2s+1} \end{bmatrix}, \quad B(s) = \begin{bmatrix} \dfrac{k_1}{T_1s+1} & 0 \\ 0 & 1 \end{bmatrix}, \quad G_0(s) = \begin{bmatrix} \dfrac{1}{\mu(T_0s+1)} & -\dfrac{P_T(T_bs+1)}{\mu(T_0s+1)} \\ \dfrac{1}{T_0s+1} & \dfrac{P_TT_ts}{T_0s+1} \end{bmatrix} \tag{12}$$

The model is simple but can capture the essential dynamics of the unit, so it can serve as a base model for controller tuning.

3 PID controller design and tuning

We see from the model in the previous section that the system is coupled only with $G_0(s)$. So, to decouple the system, we just need to decouple this part. Note that the inverse of G_0 is

$$G_0^{-1} = \begin{bmatrix} \mu T_ts & T_bs+1 \\ -\dfrac{\mu}{P_T} & \dfrac{1}{P_T} \end{bmatrix} \begin{bmatrix} \dfrac{T_0s+1}{(T_b+T_t)s+1} & 0 \\ 0 & \dfrac{T_0s+1}{(T_b+T_t)s+1} \end{bmatrix} \tag{13}$$

Clearly, a decoupler can be chosen as

$$W_0 = \begin{bmatrix} \mu T_ts & T_bs+1 \\ -\dfrac{\mu}{P_T} & \dfrac{1}{P_T} \end{bmatrix} \tag{14}$$

Now, a decoupler for the whole unit (without considering its realizability at present) can be chosen as

$$\tilde{W}_0 = \begin{bmatrix} \dfrac{T_1s+1}{k_1} & 0 \\ 0 & 1 \end{bmatrix} W_0 = \begin{bmatrix} \dfrac{\mu T_ts(T_1s+1)}{k_1} & \dfrac{(T_1s+1)(T_bs+1)}{k_1} \\ -\dfrac{\mu}{P_T} & \dfrac{1}{P_T} \end{bmatrix} \tag{15}$$

To make it realizable and consider the requirement of setpoint tracking, we cascade it with integrators. Thus the final decoupler is

$$W_1 = \tilde{W}_0 \begin{bmatrix} \dfrac{1}{s} & 0 \\ 0 & \dfrac{1}{k_2s} \end{bmatrix} = \begin{bmatrix} \dfrac{\mu T_t(T_1s+1)}{k_1} & \dfrac{(T_1s+1)(T_bs+1)}{k_1k_2s} \\ -\dfrac{\mu}{P_Ts} & \dfrac{1}{P_Tk_2s} \end{bmatrix} \tag{16}$$

Now, we have a PID decoupler. With this decoupler, the compensated model will be

$$\tilde{G} = \begin{bmatrix} \dfrac{(T_b+T_t)s+1}{T_0s+1}\dfrac{1}{s} & 0 \\ 0 & \dfrac{(\alpha T_2+1)}{T_2s+1}\dfrac{(T_b+T_t)s+1}{T_0s+1}\dfrac{1}{s} \end{bmatrix} \tag{17}$$

We see the gain of the compensated model is normalized. That is the reason why we include k_2 in the decoupler. We need to design two single-loop controllers (K_1 and K_2) for the diagonal elements of $\tilde{G}$ to improve dynamic response of the system. The final coordinated controller will be of the form

$$K = \begin{bmatrix} \dfrac{\mu T_t(T_1s+1)}{k_1} & \dfrac{(T_1s+1)(T_bs+1)}{k_1k_2s} \\ -\dfrac{\mu}{P_Ts} & \dfrac{1}{P_Tk_2s} \end{bmatrix} \begin{bmatrix} K_1 & 0 \\ 0 & K_2 \end{bmatrix} \tag{18}$$

Now, we have the controller structure for the boiler-turbine unit. To tune the controller parameters, we need six parameters $(\mu, P_T, k_1, k_2, T_b, T_t)$ from the unit model. These parameters can be obtained from the step response data of the boiler-turbine unit at the tuning operating point. Suppose a transfer function $G(s)$ is obtained by fitting the step response data for firing rate and governor valve position. Then we have

$$\frac{k_1}{\mu} = g_{11}, \ \frac{P_T}{\mu} = g_{12}, \ k_1k_2 = g_{21}, \ P_Tk_2T = \lim_{s\to 0}\frac{G_{22}(s)}{s} =: g_{22} \tag{19}$$

Where g_{11}, g_{21} denote the static gains of the (1,1) and (2,1) blocks of $G(s), g_{12}$ denotes the *absolute value* of the static gain of the (1,2) block of $G(s)$, and g_{22} denotes the rate of the (2,2) block of $G(s)$. Then T_t and $P_T k_2$ can be calculated as

$$T_t = \frac{g_{22}g_{11}}{g_{12}g_{21}}, \quad P_T k_2 = \frac{g_{12}g_{21}}{g_{11}} \tag{20}$$

T_b, T_0 can be directly estimated from the (1,2) block of $G(s)$ by approximating it with a first-order model. When T_0 is estimated, T_1 can also be estimated from the (1,1) block of $G(s)$.

Now, we need to tune K_1 and K_2 for the compensated model $\tilde{G}$ in (17). A candidate for K_1 can be chosen as

$$K_1 = \frac{T_0 s + 1}{(\lambda s + 1)((T_t + T_b)s + 1)} \tag{21}$$

However, since the time constant T_0 is usually very large for a boiler-turbine unit, this inverse-based controller needs a large initial input to achieve the desired response. So we would rather choose a simple gain, say λ_1, meaning that we require the response time be about $1/\lambda_1$.

Note that the time constant T_2 is usually small compared with T_0, and α is usually between 1/4 and 1/2. So K_2 can also be chosen as a gain λ_2, with $1/\lambda_2$ determining the desired response time. However, since we usually require that the response for the electric power demand be as fast as possible, we may choose K_2 as a PD of the form $\lambda_2(1+T_{d2}s)$ to accelerate the response and reduce the overshoot.

The final coordinated controller will have the following structure

$$K_{PID} = \begin{bmatrix} \frac{\mu T_t(T_1 s + 1)}{k_1} & \frac{(T_1 + T_b)s + 1}{k_1 k_2 s} \\ -\frac{\mu}{P_T s} & \frac{1}{P_T k_2 s} \end{bmatrix} \begin{bmatrix} \lambda_1 & 0 \\ 0 & \lambda_2(1 + T_{d2}s) \end{bmatrix} \tag{22}$$

The approximation above will ensure that each element of the controller can be realized with a PID structure. Moreover, if T_1 is very large, we may use a static gain for the (1,1) block to reduce the effect of a large derivative action.

In summary, a coordinated PID controller for a boiler-turbine unit can be tuned by the following procedure:

1) At a certain steady state, switch the firing rate control and governor valve control to manual, do step response tests for firing rate (or fuel flow) and governor valve position, respectively. Record the response data.

2) Compute g_{11}, g_{22}, g_{21} and g_{12}; compute T_t and estimate T_0, T_b and T_1.

3) Tune λ_1 for model $\frac{(T_b + T_t)s + 1}{s(T_0 s + 1)}$ to have the desired response.

4) Estimate T_2 and α from the step response data, and tune λ_2, T_{d2} for model $\frac{(\alpha T_2 s + 1)((T_b + T_t)s + 1)}{s(T_2 s + 1)(T_0 s + 1)}$ to have the desired response.

5) The final PID controller is realized by (22).

In fact, step 4 can be omitted. The reason is that when T_2 is small compared with T_0, $\frac{\alpha T_2 s + 1}{T_2 s + 1}$ will have little effect on the dynamics of $\frac{(T_b + T_t)s + 1}{s(T_0 s + 1)}$. λ_2 can then be tuned for $\frac{(T_b + T_t)s + 1}{s(T_0 s + 1)}$ and T_{d2} can be tuned to damp the overshoot and/or the coupling effect.

4 Simulation study

Consider a 500MW nonlinear dynamic model (Fig. 2) established for Shentou No.2 power plant, Shanxi, China[8]. The rated parameters are steam flow 1650t/h, drum pressure 18.97Mpa, and steam pressure 16.18Mpa.

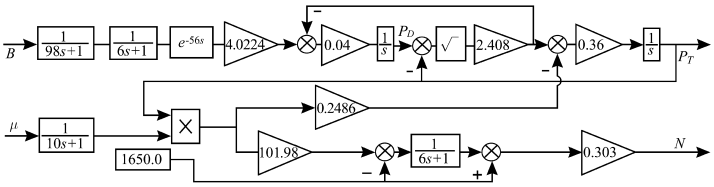


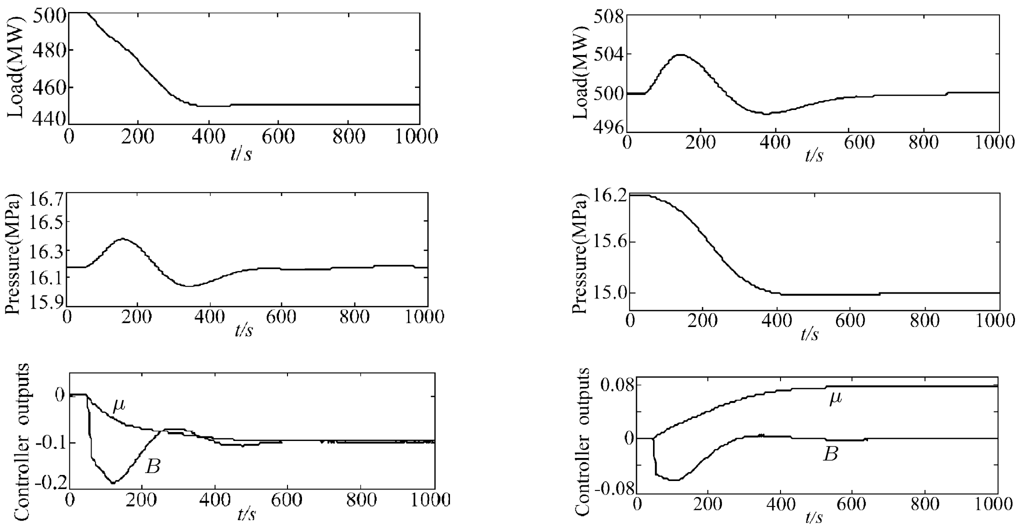
Fig. 2 Nonlinear model of a 500MW unit

From the step response at full load, typical parameters in Eq.(11) are identified as: $g_{11} = 16.2, g_{12} = 16, g_{21} = 1001.2, g_{22} = 110640$, and $T_0 = 140, T_1 = 110, T_t = 111.9, T_b = 20$. Therefore the coordinated controller has the following structure

$$K_{PID} = \begin{bmatrix} 6.91(1 + 110s) & \dfrac{130s + 1}{1001.2s} \\ -\dfrac{1}{16s} & \dfrac{1}{988.8s} \end{bmatrix} \begin{bmatrix} \lambda_1 & 0 \\ 0 & \lambda_2(1 + T_{d2}s) \end{bmatrix} \tag{23}$$

The parameters λ_1 and λ_2 are tuned to be 0.006 and 0.02, respectively, and T_{d1} is chosen as 40 to damp the oscillation caused by using a larger gain.

With the assumption that the system is in a steady state with $P_D = 18.97$MPa, $P_T = 16.18$MPa, $N = 500$MW, $B = \mu = 1$, initially, the step responses for the closed-loop system (step starts from $t = 50$) are shown in Fig. 3. Case 1 (load demand decreased from 500MW to 450MW) is shown in Fig. 3(a), Case 2 (setpoint of steam pressure decreased from 16.18MPa to 15MPa) is shown in Fig. 3(b).

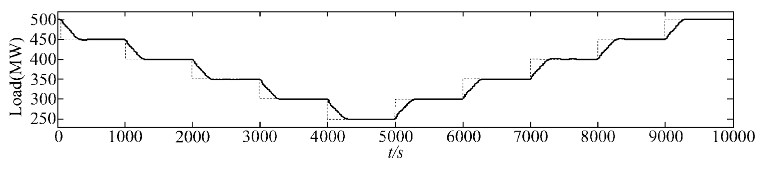


(a) Step responses in case 1

(b) Step responses in case 2

Fig. 3 Simulation for a step decrease of load and steam pressure

To test the performance for a wide range of operating points, we reduce the load demand from 500MW to 250MW with a step of 50MW, and then increase the demand step by step. The responses of load and pressure are shown in Fig. 4. It can be seen that the controller has good performance for this range of load demand.



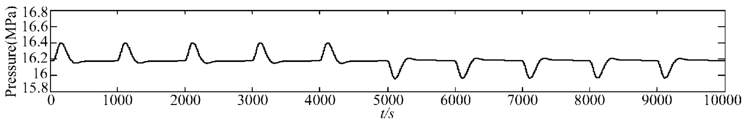


Fig. 4 Simulation of electric load demand decreasing from 500MW to 250MW and then going back

5 Conclusions

A simple model for a boiler-turbine unit is derived in the paper and based on this typical model a PID tuning method for the coordinated controller is proposed. Example shows that the method is easy to use and can achieve good performance. Further research might be done to improve the overall performance, These including

Modeling. A more sophisticsted model for a boiler-turbine unit may give a better control structure than the one given in the paper.

Control structure for large time constant. Since T_0 is usually very large for a unit, to improve the performance we may expect structures other than PID.

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